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(71)(72) Applicant and Inventor: ADORJÁN, Ferenc [HU/HU]; Csipke u. 1/b, H-1125 Budapest (HU).

(74) Agent: FABER, Miklós; Advopatent, Fő u. 19, H-1011 Budapest (HU).

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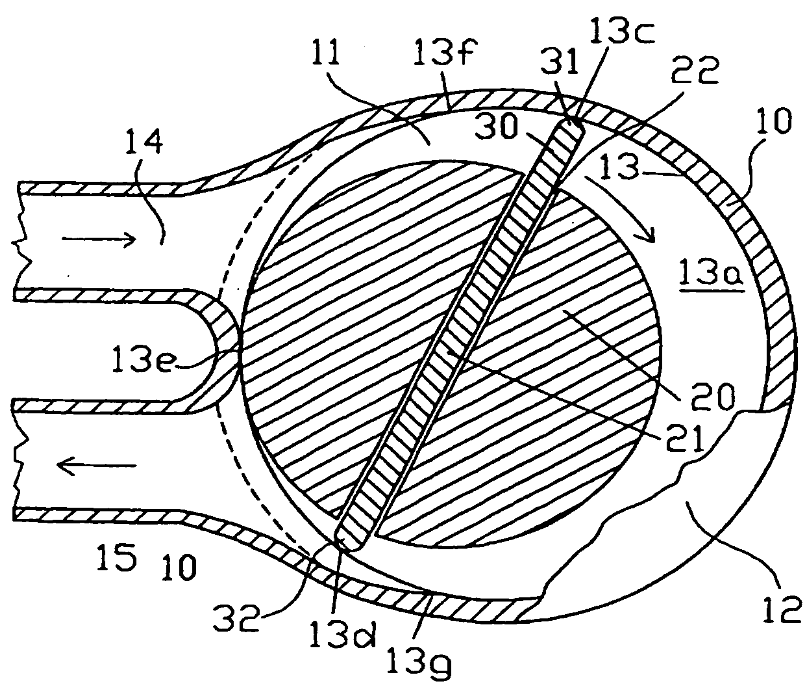
(54) Title: ROTARY ASSEMBLY

(57) Abstract

A rotary assembly consisting of a housing (10) with a cavity (13a) that is bounded by the first and the second planar walls (11, 12) as well as by a curved, closed surface (13) in-between, with inlet (14) and outlet (15) openings connecting to the cavity, the rotatably mounted shaft (20) with an axis perpendicular to the planes of planar walls is furnished by a slot (22), parallel and symmetrical to the axis of the shaft and limited within the section of the shaft inside the parallel walls of the cavity and with one or more vanes (30) fitted into the slot which divide the cavity into two or more sub-sections.



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Rotary assembly

The different types of sliding vane rotary pumps used for liquid or gaseous substances, as well as the oil-sealed vacuum pumps have been widely applied and rather well known for long time. Typically, they are built with internal cavities bounded by cylindrical and planar surfaces and their shaft is positioned eccentrically. The vanes that are fitted into appropriate slots of the shaft are forced to touch the cylindrical surface of the cavity by using springs or by centrifugal force. The traditional such assemblies share the disadvantages that a substantial friction arises along the contact lines of the vanes with the internal surface of the cavity, while causing direct limitation on the head (pressure drop) of the pump. In case of the pumps with vanes positioned by centrifugal force, a minimum rotational speed is also required before the pump would work.

A further limitation for such pumps (or hydraulic motors) is that in case of operating with incompressible fluids the volume in the slots of the shaft between or behind the vanes (spring space) changes during the operation of the assembly which requires passageways or loose fitting of the vanes into the slots, which deteriorates the volumetric efficiency. All such kind of pumps generate a more or less oscillating fluid flow that oscillation can only be lowered by increasing the number of vanes that causes friction penalty.

Some of the above described disadvantages are eliminated by construction described in the US patent No. 5.006.053. The essence of this is that the cross section of the internal curved surface of the pump cavity follows a given function (in fact the Pascal's limaçon curve with the

eccentricity parameter of 0,5) and there is a single vane through the slot of the shaft. The main limitation of this construction is that the proposed surface geometry does not ensure the exact continuous contact between both ends of the vane and the curved surface, unless the vane is infinitely thin. The oscillation of the pump flow is not eliminated at all. These conditions pose serious limitations for the applicability of such pumps (e.g. for precision applications).

There exists at least one known proposal to apply the eccentric rotary assembly with sliding vanes as an internal combustion engine: e. g. the equipment as described in the US patent No. 4,909,208. This construction, however, has the potential shortcomings both from point of view of increased friction due to the spring supported vanes and from difficulties in sealing the combustion chamber. These problems are anticipated, since the sudden pressure peak following the ignition could push the vane easily into the slot, creating a free escape gap for the hot gases, unless the vane springs are extremely stiff.

The current invention is aiming at eliminating the above described shortcomings of the known sliding vane rotary assemblies by introducing such a general geometrical solution that ensures the reliable close contact between the vane and the internal surface of the cavity regardless of the rotational speed and without creating substantial friction forces. Under some special conditions the proposed equipment produces (or in case of a hydraulic motor, it requires) oscillation free fluid flow with constant rotational speed of its shaft. When the proposed arrangement is applied as the compression and expansion units of an internal combustion engine, the resulted equipment is free from many difficulties that arise in case of the formerly proposed rotary combustion engines.

The solution proposed in the current invention is based on the geometrical discernment that the locus of the endpoints of a straight section which rotates 180 degrees in the plane around

a fixed point such a way that said point stays on the section without being fixed to any given point of the section, obviously exists and can easily be created. Any plane curve that has been derived such a way share the common feature that at least one of the cords in any given direction which go through the selected fixed point (the cord centre) has the same length as the original section. (Note that for convex curves - and only those are relevant from the point of view of the invention - there is only one single cord of the curve in any direction.) Later on, we will refer the curves that belong to this very generally defined plane curve family as equicord curves. Thus, the $r(\varphi)$ functional form of such curves in (r,φ) polar co-ordinates centred at the cord centre, satisfy the equation

$$r(\varphi) = 2c - r(\varphi + \pi)$$

where $2c$ is the length of the cords which lie on the cord centre (the angle is measured in radians). It is obvious that any of such closed curves can freely be defined within an angle range of $\varphi \in [0,\pi]$ and the other half of the curve comes from the above equation. This option allows extremely high freedom in creating specific forms of such curves. A great variety of different plane curves, even curve families satisfy the above equicord condition, from among those the best known is the Pascal's limacon curve (with some restrictions on its parameters).

The above outlined geometrical features of the equicord curves got materialisation in the invention by discerning that a single, rigid vane can only touch by both of its rounded ends the internal curved surface of the assembly cavity - regardless of its angular position - if the cross section of the cavity is derived from an equicord curve and the rotational axis goes through the equicord point. It can be proven that if in cross section perpendicular to the rotational axis the ends of the vane follow circular convex arcs then the cross section of the required shape of the cavity has to correspond to the outer parallel of an equicord curve at the distance of the

rounding radius of the vane. We obtained such a way a generic geometry for a rotary assembly, assuming appropriate shaft geometry.

Corresponding to the objectives of the invention, the rotary assembly - consisting of a housing with a cavity that is bounded by the first and the second planar walls, as well as by a curved, closed surface in-between, with inlet and outlet openings connecting to the cavity, the rotatably mounted shaft with an axis perpendicular to the planes of planar walls, furnished by a slot parallel and symmetrical to the axis of the shaft and limited to the section of the shaft inside the parallel walls of the cavity; with one or more vanes fitted into the slot which divide the cavity into two or more sub-sections, - is designed such a way that the end surfaces of the vane (vanes) toward the curved surface of the cavity follow rounded curves in cross section parallel to the planar walls of the cavity, while in any such cross section of the curved surface of the cavity follows a non-circular curve, having a primary internal fitting point which lies on the rotational axis of the shaft and this non-circular curve is determined such a way that the rounded end surfaces of the vane are in fluid-tight contact along two opposite generatrices with the curved surface of the cavity regardless the angular position of the shaft; the shaft and the internal curved surface of the cavity are also in a permanent fluid-tight contact at least along a single fixed generatrix or a range of generatrices of the curved surface; while the inlet and the outlet openings are on the two sides of this contact range.

A preferred embodiment of the rotary assembly is characterised by identical circularly rounded end surfaces of the vane and the cross section of the curved surface of the cavity follows a geometrical curve which is an outer parallel of an equicord curve (satisfying the equation $r(\varphi)=2c-r(\varphi-\pi)$ in polar co-ordinate system centred at the primary fitting point of the curve) at the distance of the rounding radius of the vane.

It is also a preferred case when the vane is a solid, single piece component.

In some other preferred embodiments of the rotary assembly there are at least two separate cavities near to each other with separate vanes in them (twin cavity arrangement).

In case of a preferred twin cavity arrangement the vanes are fitted into the slots of the same shaft that traverses both cavities.

A further twin cavity arrangement is characterised with identical cavities, with vanes in them perpendicular to each other in projection to a plane perpendicular to the shaft axis and the inlets of the cavities divide from a common inlet, as well as the outlets of the cavities merge into a common outlet.

In case of another preferred dual cavity arrangement the outlet of the first cavity in its housing is connected to the inlet of the second cavity and there is a movable valve body in the connecting channel that operates in a synchronised way with the single shaft or the shafts that encompass the sliding vanes; and there exists - preferably at one side of the valve - a fuel injection nozzle and an igniting equipment.

The invention exhibits several advantageous features. Perhaps the most significant such feature is that it is possible to create a pump from it that is applicable for carrying gaseous, liquid or mixed fluids and which is characterised by high volumetric efficiency, high head, low energy loss through friction between the cavity wall and the vane; which not only improves its energy efficiency but also increases the lifetime of the assembly. The same or a similar assembly can also be applied as a hydraulic motor having analogous beneficial qualities.

It is a general advantageous peculiarity of the invention that the constraints on the shape of the curved surface of the cavity allow extremely high variability and continuous parameter

adjustment that opens a broad range of options for the designers to optimise the equipment to satisfy a wide range of specific requirements.

It is of high significance that in case of the dual cavity arrangement with perpendicular vanes, the sum of the volumetric flow rate of the two chambers will be constant (oscillation free) if the shape of the internal cavities are determined suitably. This is a characteristic case of satisfying an extra condition by designing the shape of the internal cavity. When such an assembly is used in hydraulic motor mode with incompressible fluid then it will also create constant torque at any angular position of the shaft.

The lack of oscillation in the flow rate together with the high volumetric efficiency opens a new, wide range of applicability ranging from precision measuring and analytical equipment to hydraulic transmissions.

By adding appropriate auxiliary devices to it, such as fuel injection and ignition devices, it is possible to create a rotary internal combustion engine which could presumably be tuned to produce extraordinary efficiency with a wide range of different fuels.

It is an additional remarkable advantage of the invention that it consists of a small number of parts which can be manufactured and assembled by using commonly applied technology.

It is further notable advantage that by combining several such assemblies into a hydraulic or pneumatic circuit many useful applications can be obtained, such as multistage torque converter, differential driving mechanism etc. In many of such combinations there is no need for expensive additional equipment (e.g. clutch), it shows, however, remarkable improvements in comparison to the traditional solutions.

In the followings the invention will be described by using figures that present explanatory sketches and sketches of different preferred embodiments. The brief description of figures is:

Figure 1 Side view cross section of a basic arrangement of the rotary assembly.

Figure 2 Sketch of a possible geometry for deriving the cross section of the shape of the internal cavity.

Figure 3 The geometry of the touching point between the cavity surface and the rounded end surfaces of the vane.

Figure 4 Section view of the oscillation free pump or hydraulic motor (corresponds to the IV-IV plane as shown in Fig. 5).

Figure 5 Section corresponding to V-V plane in Fig. 4.

Figure 6 Hydraulic driving circuit built from oscillation free pump and hydraulic motor in active state.

Figure 7 The same circuit as in Fig. 6 in neutral state.

Figure 8 Hydraulic differential driving

Figure 9 Scheme of a multistage torque converter built from oscillation free assemblies.

Figure 10 Internal combustion engine built from two rotary assemblies in the moment of valve opening.

Figure 11 The same engine in the moment of valve closing.

Fig. 1 shows a generic arrangement of the rotary assembly with its housing 10, its first planar wall 11 and its specially curved surface 13 enclosing the cavity 13a. The inlet 14 and the outlet 15 openings are connecting to the cavity 13a. The position of the shaft 20 within the cavity 13a of the housing 10 is well shown. The slot 22 traversing the shaft 20 holds the (in this typical case single piece) vane 30 having the rounded end surfaces 31,32. The rotary shaft 20 is

connected to the planar walls 11.12 of the housing 10 by appropriate bearing such a way that the projection of the primary fitting points of the curved surface of the housing should fall onto the rotational axis 21 of the shaft.

As it can be observed in Fig. 1, the surface of the shaft 20 and the curved surface 13 of the housing 10 has connecting generatrices at the point 13e. Along these generatrices which are parallel to the rotational axis 21 of the shaft 20 the two surfaces are in fluid tight contact. The same fluid tight contact is maintained at the points 13c and 13d where the generatrices of the rounded end surfaces 31.32 of the vane 30 and the curved surface 13 of the housing 10 are in contact. The inlet 14 and the outlet 15 are always separated from each other by at least one of the generatrices 13c and 13d during the rotation of the shaft 20.

It is also clear form Fig. 1 that the inlet 14 and the outlet 15 openings are situated on the two sides of the connecting generatrix 13e of the surface 13 and the surface of the shaft 20. The inlet 14 and the outlet 15 openings extend down to the zenith 13f and nadir 13g generatrices of the curved surface 13. respectively. These generatrices are defined by the touching lines between the vane 30 and the surface 13 when the vane 30 is in upright position (perpendicular to the plain determined by the axis 21 and generatrix 13e). Should the inlet 14 and the outlet openings extend further than the generatrices 13f and 13g, the full separation of the input and the output volumes could not be maintained.

It is necessary to note here that when the assembly is designed for using with incompressible fluid the extension of the inlet 14 and outlet 15 openings to the zenith 13f and the nadir 13g generatrices is a necessary requirement. The inlet 14 and outlet 15 openings have to be created such a way that the curved surface 13 of the housing 10 should go continuously around the openings to control the motion of the vane while the shaft 20 rotates around. The inlet 14

and outlet 15 openings can be designed partially or fully on the planar walls 11,12 of the housing by taking into account the above described limitations.

During the pump operational mode of the assembly shown in Fig. 1. the shaft 20 rotates in the direction as indicated by the arrow and the vane 30 rotates with it while its rounded end surfaces 31,32 follow the curved surface 13 of the cavity 13a of the housing 10. At the moment when the end surface 31 of the vane 30 reaches the generatrix 13e of the 13 surface. the space above the vane is in connection with the volumes linked to the inlet opening 14 (suction or low pressure side): while the space below the vane 30 is in connection with the volumes linked to the outlet opening 15 (high pressure side). The two sides are separated by the section of the vane 30 outside the slot 21 of the shaft 20 in the direction of the end surface 32, as well as the connecting generatrix 13d. Later on but until the vane 30 would reach the upright (90 degree) position the separating surfaces remain the same, since the connection between the rounded surface 31 of the vane at the generatrix 13c is broken, assuming that the inlet opening extends from the generatrix 13e to the zenith line 13f. During this phase the volume of the suction side increases and the volume of the high pressure side decreases continuously. At the moment when the vane 30 reaches the upright position. when the end surface 31 touches the surface 13 along the zenith generatrix 13f, the volume on the right side of the vane 30 is cut away from the suction side and for a moment there is a confined volume between the generatrices 13f and 13g. As the shaft 20 rotates further the confined volume will be connected to the high pressure side (assuming that the outlet opening 15 extends to the generatrix 13g) and the role of separation is taken over by the generatrix 13c and the end section of the vane 30 toward its end surface 31. As the vane moves the suction side expands further continuously and the formerly confined volume on the right side of the vane 30 is being displaced continuously toward the high pressure side. During a full rotation of the shaft 20. the above process takes place two times.

Fig. 2 presents a geometrical sketch of the cross section of the internals of the assembly (perpendicular to the rotational axis of the shaft) showing as the symmetry line 30a of the vane 30, which goes through the primary fitting point "O", and as the arched contour lines 31 and 32 of the vane are in contact with the 13b bounding curve (corresponding to the surface 13 in Fig. 1) on the opposite sides of point "O", regardless the angular position of the vane. The thin curve 133 is the equicord curve from which the bounding curve 13b has been derived. The actual shape of the curves correspond to the shape of the cavity of the twin arrangement designed to provide constant flow, as described below.

Fig. 3 shows in an enlarged scale how the bounding curve 13b can be derived from the equicord curve 133 when the end surfaces 31,32 of the vane 30 are circular arches in cross section with radius R. This derivation is necessary, since the equicord curve 133 could secure the exact connection between the vane and the curved surface of the cavity only if the thickness of the vane is zero. Thus, taking into account the real measures of the vane 30, we need to substitute the original curve 133 by the outer parallel curve at the distance of R. An arbitrary point of this parallel can be obtained (as it is well known) if we take the outer normal vector of the point "E" of the original curve and measure the distance "R" to it, obtaining point "X". These points generate the bounding curve 13b. Note that if the symmetry line 30a of the cross section of the vane goes through point "E" then the touching point between the cross section of the rounded end surface of the vane and the bounding curve 13b is at "X". If the rounding of the end surfaces of the vane 30 are non-circular or non-symmetric then the derivation of the bounding curve from the equicord curve is still possible but different and more complicated.

Figures 4 and 5 present a rotary assembly having two internal cavities 13a and 131a within its housing 10, which are identical in shape and size. In this twin or dual construction the vane 30 in the cavity 13a and the vane 301 in the cavity 131a are also identical and they are

fitted into the slots of the same shaft 20. It is important that in perpendicular projection in the direction of the rotational axis 21 of the shaft 20 the angle α between the two vanes 30, 301 is 90 degree. At this variant of the assembly the inlet 16 of the assembly is connected to both of the cavities 13a and 131a through the openings 14 and the exit openings 15 of the cavities 13a and 131a merge into the common outlet 17.

Since this arrangement is mostly useful when it operates with some incompressible fluid the inlet openings 14 and the outlet openings 15 should extend from the vicinity of the generatrix 13e up to the zenith 13f and down to the nadir 13g generatrices, respectively. The inlet 14 and the outlet 15 openings of the cavities 13a and 131a may extend partly or fully over the planar walls 11, 12 and 12a of the cavities assuming that they do not extend over the plain determined by the zenith 13f and the nadir 13g generatrices.

It is of high importance that the curved surfaces of the cavities of the twin assembly can be chosen such a way that when the assembly is applied as a pump with some incompressible fluid, then the volume of the displaced fluid will be strictly proportional to rotational angle of the shaft 20 (up to the accuracy of the 13 surface of the cavities and the other internal components). If this twin arrangement operates in hydraulic motor (turbine) mode then the rotational angle of the shaft 20 will be proportional to volume of the transmitted fluid. In other words, we obtained a pulsation-free rotary pump or hydraulic motor. Such pumps and hydraulic motors are well applicable in every case when well controlled volumetric flow or rotation with significant pressure drop or torque is required. From among the great variety of possible applications a few is described below.

In Figs. 6 and 7 a hydraulic circuit is presented which consists of two twin rotary assemblies: a pump 71 and a motor 72. Excluding the high pressure line 76 and the low pressure

line 77. the other auxiliary elements of the circuit are application dependent, thus optional. It follows from the facts described above that the rotation of the shaft of the motor 72 is proportional to the rotation of the shaft of the pump 71 and the transmission ratio is determined by the volume ratios (and secondarily the slip) of the two assemblies.

The buffer volume 73 may be connected to any point in the circuit and it is required only if the circuit is closed and the heat expansion of the working fluid significantly differs from that of the solid components. If some extent of elasticity is beneficial from the point of view of the application then it should be connected to the high pressure line 76.

When the circuit is completed with the three-way two-position routing valve 74 and the shorting line 78, then in the state corresponding to Fig. 6 the torque transmission will take place, while in the state as shown in Fig. 7 the pump runs in idle mode. In the working state of the circuit the routing valve 74 leads the fluid from the outlet of the pump through the high pressure line 76 toward the inlet of the motor 72, while the shorting line 78 is closed. In idling mode the high pressure line 76 is closed and the fluid from the outlet of the pump is directly recycled through the shorting line 78 toward its inlet. If the routing valve is of overlapping type then in its middle state a fraction of the fluid flow will go toward the line 76 while the other part of the fluid goes through the line 78, ensuring a smooth transition of the circuit from one state to the other.

By adding a further shorting line 79 to the circuit that contains the check valve 75, one obtains a free wheel driving arrangement. Whenever due to either some outer force or its inertia the shaft of the motor 72 would rotate faster than it is determined by the transmission ratio of the circuit, the check valve 75 opens letting through the excess flow produced by the motor 72 in pump mode. If the flow through the check valve 75 can be controlled from outside then the

overrun of the shaft of the motor 72 can be controlled or braked. In the state of the circuit corresponding to Fig. 7, when the pump 71 is idling, the full range braking of the shaft of the motor 72 can be realised.

Fig. 8 presents a hydraulic circuitry containing three twin chamber rotary assemblies;: one 71 working in pump mode and the other two 72, 72a working in motor mode. The high pressure line 76 branches symmetrically into the lines 76a and 76b, which lead to the inlets of the hydraulic motors 72 and 72a. The outlets of the motors 72 and 72a merge into the low pressure line 77, connected to the inlet of the pump 71. The torque produced on the shafts of the two driven motors 72,72a is equal to each other and is proportional to the pressure difference between the high pressure 76 and the low pressure 77 lines. The sum of the rotations of the shafts of the two driven motors 72,72a is proportional to the rotations of the driving pump 71. In summary, the behaviour of this circuit very well corresponds to the classical planetary gear differential driving mechanism.

When the circuit in Fig. 8 is completed with the routing valve 74a, the overrun controlling check valves 75a and 75b, and the buffer volume 73, then the features as described above can be added to the circuit.

We obtain a rather specific kind of differential driving mechanism by adding the valve 81 to the circuit. The function of this valve is to balance the flow difference between the branches 76a and 76b through a negative feedback. (The details of such a valve do not belong to this invention.) The advantage of applying such a flow balancing valve 81 is that it solves the inherent shortcoming of the mechanical differential drives, the twirling that occurs when the braking torque on one of the shafts of the motors 72,72a is much lower than on the other shaft. The flow balancing valve tends to decrease the flow rate in the branch passing a higher

volumetric flow (and as a result decreasing the torque on the shaft of that motor) while increases the flow rate in the other branch, causing a higher torque on the shaft of the motor on this side. Such a feature can be created with the mechanical differential drives only in a very complicated way though it is very beneficial for most vehicle drives.

A further feature of this kind of hydraulic differential drive is that - in contrary to the traditional mechanical drives - it causes no trouble at all to drive more than two motors in differential mode by creating multiple branches. If a circuit with stepwise binary branching is applied, the flow balancing valve can also be applied. Such a driving mechanism could especially be advantageous for vehicles designed for heavy terrain with four or more driven wheels.

If one modifies the circuit shown in Fig. 8 such a way that it will contain on its driving side multiple pumps like 71, so that they are mounted on a single shaft, then a multiple stage torque converter is obtained which is well suitable for automatic control. Such a scheme is shown in Fig. 9. containing three driving side pumps, 71, 71a and 71b. The common shaft 91 of the pumps can be driven e.g. by an internal combustion engine. The driven side as shown in Fig. 9 is a differential driving circuit - corresponding to Fig. 8 - though it could be any sub-circuitry containing some hydraulic motors. The outlets of the pumps 71, 71a, and 71b are connected to the volume 92 which leads the fluid further through the high pressure line 96. The role of the volume 92 is no more than ensuring a low resistance connection between its inlets and its outlet. The fluid returning from the driven side enters the volume 93 that divides it toward the inlets of the driving pumps 71, 71a, and 71b. The transmission ratio between the shafts of the driving and the driven side can be changed in this hydraulic circuit by using the routing valves 74b, 74c, and 74d. When all the three routing valves are in idling state (as the state of the valve 74c in Fig. 9), then the whole transmission circuit is in idle (neutral) state. When the pump having the smallest volume (e.g. the pump 71b with the valve 74d) is switched on then the transmission provides the

highest transmission ratio, analogue to the lowest gear with mechanical transmissions. When this pump switched off and the pump next in volume (e.g. pump 71a) is switched on simultaneously, then the second transmission ratio can be set. The next transmission ratio can be obtained for example keeping the pump 71a on and switching on the pump 71b, as well. The lowest transmission ratio - corresponding to the highest gear - is set when all the pumps 71, 71a, and 71b are in on state. It is a notable feature of this circuit that it does not require any kind of clutch for switching from one state to the other, assuming that the routing valves have such intermediate position when both of the controlled branches are partially open.

The number of different transmission ratios of such a circuit depends on the number of pumps in the circuit and the relative volume ratios of the pumps. When the volume ratios are well chosen then with two pumps (71, 71a) three transmission ratios can be obtained, with three pumps (71, 71a, and 71b) seven and with four pumps 16 different ratios can be obtained.

If one out of the driving side pumps 71, 71a, and 71b is connected into the circuit in opposite way compared to the others (i.e. its inlet is connected to volume 92 and its outlet to volume 93) then a reverse stage of the transmission is created. If the reversed pump has the smallest volume (e.g. 71b) then it can be turned on also in the forward stages of the transmission, thus increasing the number of different forward stages. In this way the transmission can have one reverse and six forward stages.

In Figures 10 and 11 the principle and the operation of an internal combustion engine is presented. The construction shown here consists of two rotary assemblies; both are analogue to one shown in Fig. 1. and are situated in the housings 10 and 100, respectively. The assembly in the housing 100 with its shaft 201, vane 301, and its cavity 131a serves as the combustion chamber, while the assembly in the housing 10 with its shaft 20, vane 30, and its cavity 13a

serves as the compressor. The outlet 15 of the compressor cavity 13a and the inlet 141 of the combustion cavity 131a are connected to each other through the passage channel 18 that can be closed and opened by the valve body 40, which is movably mounted to the housing 10 and/or the housing 100. It seems advantageous if this valve body has a disc like shape.

One can also notice in Fig. 10 that there are the fuel injection nozzle 50 and the ignition equipment (spark plug) 60 in the passage channel 18, in this case both are on the side of the cavity 131a. The synchronised rotation of the shafts 20 and 201, as well as the valve body 40 is ensured by appropriate gearing, which is not shown in the drawings. The detailes of this gearing, as well as of the necessary synchronising tools for the fuel injection and the ignition devices do not belong to the circle covered by the invention, since many traditional solutions are available for the task.

The shafts 20 and 201, as well as the shaft of the valve 40 should rotate exactly at the same speed, keeping their relative rotational angles fixed. These relative angles has to be determined such a way that right after the moment when the edge of the vane 301 of the combustion chamber 131a has left the rim of the connecting passageway 18, the opening 41 of the valve body 40 opens to let through the compressed air from the cavity 13a toward the expanding volume of the combustion cavity 131a. In this very moment the vane 30 in the housing 10 is approaching the rim of the outlet opening 17, but it does not reach it yet. Fig. 10 represents this position of the vanes 30, 301 and the valve 40.

As the shafts rotate further and a part of the compressed air has already flown into the combustion chamber, the fuel injection nozzle injects the necessary amount of fuel into the combustion chamber 131a. Right before the edge of the vane 30 would pass the rim the outlet opening 17, the valve body 40 closes the passageway 18, confining the compressed air-fuel

mixture in the cavity 131a and the ignition device 60 ignites the mixture upon this moment, starting the expansion, working phase. Around this time inside the other end of the vane 30 in the compression chamber 13a confines a new volume of air and starts compressing it; while simultaneously on the other side of the vane toward the inlet opening 14 a new sucking phase is started. This moment is presented in Fig. 11.

Following this moment the expansion phase lasts for about half a rotation of the shafts (180 degree or somewhat more) until the edge of the of the vane 301, that confines the expanding volume reaches the rim of the exhaust opening 151. The exhaust phase goes in parallel with the next working phase on the other side of the vane 301. Thus, during each full rotation of the main shaft 201 two working phases take place, just like in case of a traditional four cylinder reciprocating engine.

The necessary lubrication of the moving parts of the engine can be solved preferably by pressing the lubricant through appropriate axial passageways within the shafts 20 and 201 toward the long edges of the vanes, which will distribute the lubricant with their movement within the cavities 13a and 131a, respectively.

As an alternative solution for this internal combustion engine the shafts 20 and 201 can be created as a single piece. This form is in fact a more preferred embodiment, though it is more difficult to present in a drawing. In this variant the passage channel 18 can be manufactured into the planar separating walls between the two chambers, and the disc-like valve body 40 could be mounted on the common shaft, as well. This construction offers several advantages over the one shown in Figures 10 and 11: no synchronisation equipment is required between the shafts 20 and 201, the valve body 40 can either be mounted on the common shaft or its driving gearing is more simple due to the parallel shafts; thus the construction is more compact and efficient.

Either the arrangement shown in Figs. 10 and 11 or its alternative single shaft form can be multiplied, e.g. if two such assemblies are mounted on a single main shaft such a way that the vanes 301 of the two combustion chambers are perpendicular to each other then an engine analogue to an eight cylinder reciprocating combustion engine is obtained. It is also important to emphasise that the restrictions on the shape of the cavity of the compression and the combustion chambers allow a great degree of freedom to optimise the expansion characteristics of the engine corresponding to virtually any kind of fuel.

The rotary assembly according to this invention can be well applied whenever high performance positive displacement pumps or hydraulic motors are required or pulsation free fluid flow or constant torque and rotational speed hydraulic motor is required. When several of such assemblies are applied in a hydraulic circuit, different torque transmission devices can be created that show several beneficial features above the traditional mechanical solutions. By combining two assemblies a well tuneable, high performance rotary internal combustion engines can be created.

THE CLAIMS:

1. A rotary assembly consisting of a housing with a cavity, that is bounded by the first and the second planar walls, as well as by a curved, closed surface in-between, with inlet and outlet openings connecting to the cavity, the rotatably mounted shaft with an axis perpendicular to the planes of planar walls, furnished by a slot, parallel and symmetrical to the axis of the shaft and limited within the section of the shaft inside the parallel walls of the cavity; with one or more vanes fitted into the slot which divide the cavity into two or more sub-sections; which is characterised by that the end surfaces of the vane (vanes) (30) toward the curved surface (13) of the cavity follow rounded curves (31, 32) in cross section parallel to the planar walls (11,12) of the cavity, while in any such cross section of the curved surface (13) of the cavity follows a non-circular curve (13b), having a primary internal fitting point (O) which lies on the rotational axis (21) of the shaft (20) and this non-circular curve (13b) is determined such a way that the rounded end surfaces (31, 32) of the vane (30) are in fluid-tight contact along two opposite generatrices (13c, 13d) with the curved surface (13) of the cavity (13a) regardless the angular position of the shaft (20); the shaft (20); the shaft and the internal curved surface (13) of the cavity are also in a permanent fluid-tight contact at least along a single fixed generatrix (13e) or a range of generatrices of the curved surface (13); while the inlet (14) and the outlet (15) openings are on the two sides of this contact range (13e).

2. A rotary assembly according to Claim 1 which is characterised by identical circularly rounded end surfaces (31a, 32a) of the vane (30) and the cross section of the curved surface (13b) of the cavity follows a geometrical curve which is an outer parallel of an equicord curve - i.e. satisfying the equation $r(\varphi)=2c-r(\varphi-\pi)$ in polar co-ordinate system centred at (Q),

equivalent to the primary fitting point of the curve (O) - at the distance of the rounding radius (R) of the vane (30).

3. A rotary assembly according to Claims 1 or 2 which is characterised by that the vane (30) is a solid, single piece component.

4. A rotary assembly according to Claims 1, 2, or 3 which is characterised by a housing (10) that contains at least two separate cavities (13a, 131a) near to each other with separate vanes (30, 301) within them.

5. A rotary assembly according to Claim 4 which is characterised by that the vanes (30,301) in the cavities (13a,131a) are fitted into the slots of the same shaft (20) that traverses both cavities (13a,131a).

6. A rotary assembly according to Claim 4 or 5 which is characterised by identical cavities (13a,131a), with vanes (30,301) in them perpendicular to each other in projection to a plane perpendicular to the shaft (20) axis (21) and the inlets (14,141) of the cavities (13a,131a) divide from a common inlet (16), as well as the outlets (15,151) of the cavities merge into a common outlet (17).

7. A rotary assembly according to Claim 4 or 5, which is characterised by that the outlet (15) of the first cavity (13a) in its housing (10) is connected to the inlet (141) of the second cavity (131a) and there is a movable valve body (40) in the connecting channel (18) that operates in a synchronised way with the single shaft (20) or the shafts (20,201) that encompass the sliding vanes (30,301); and there exists - preferably at one side of the valve (40) - a fuel injection nozzle (50) and an igniting equipment (60).

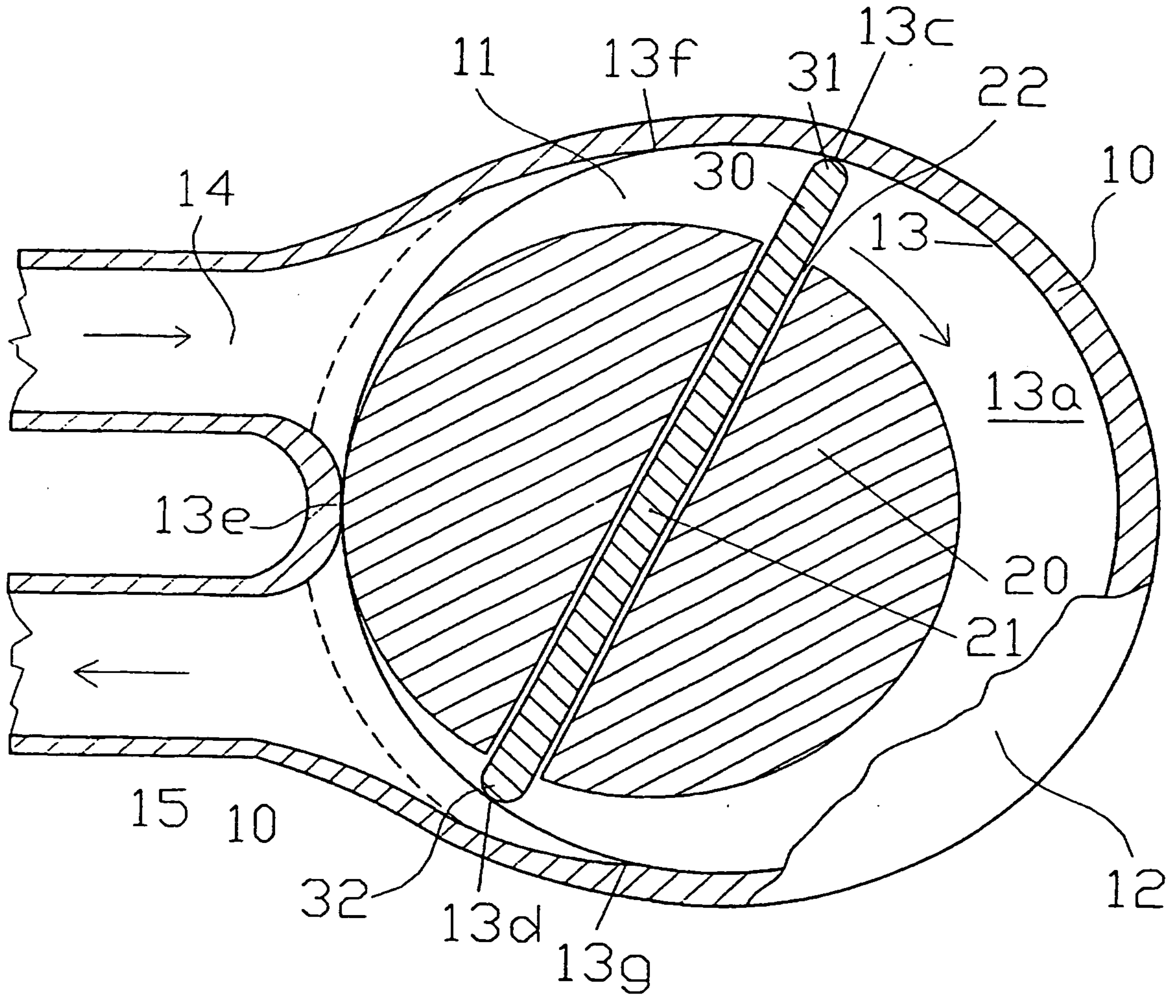

Figure 1

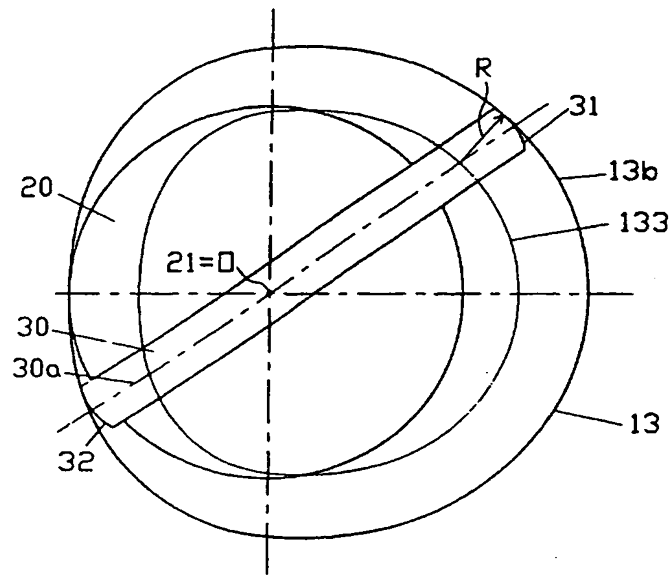

Figure 2

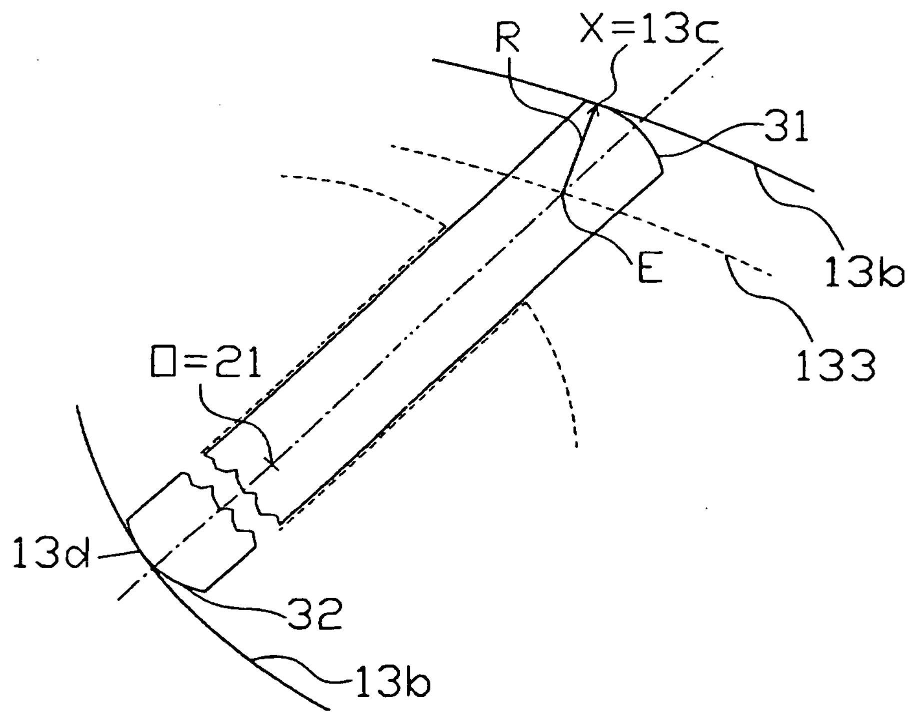

Figure 3

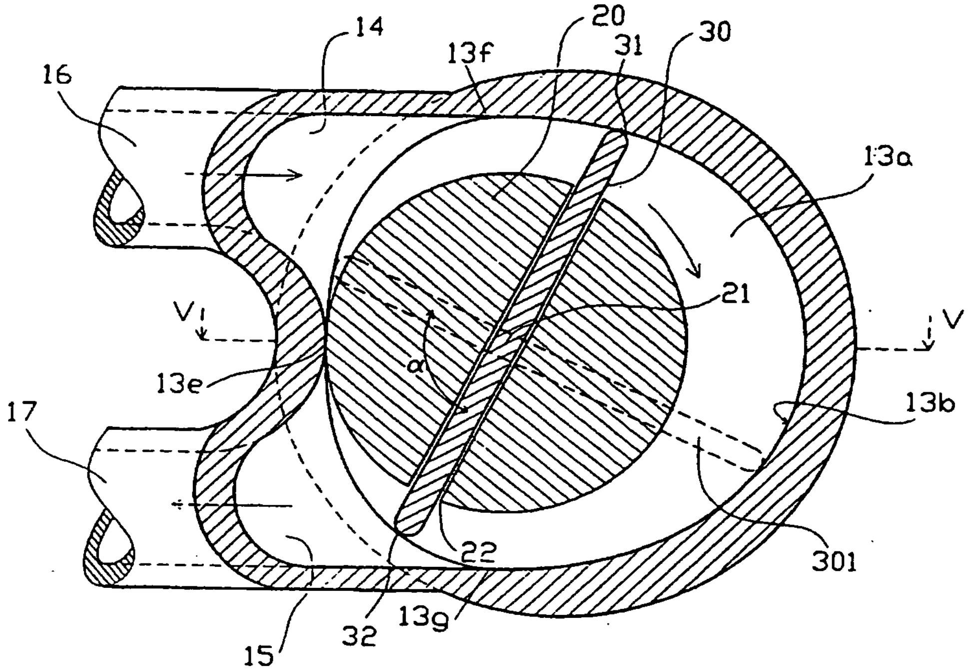

Figure 4

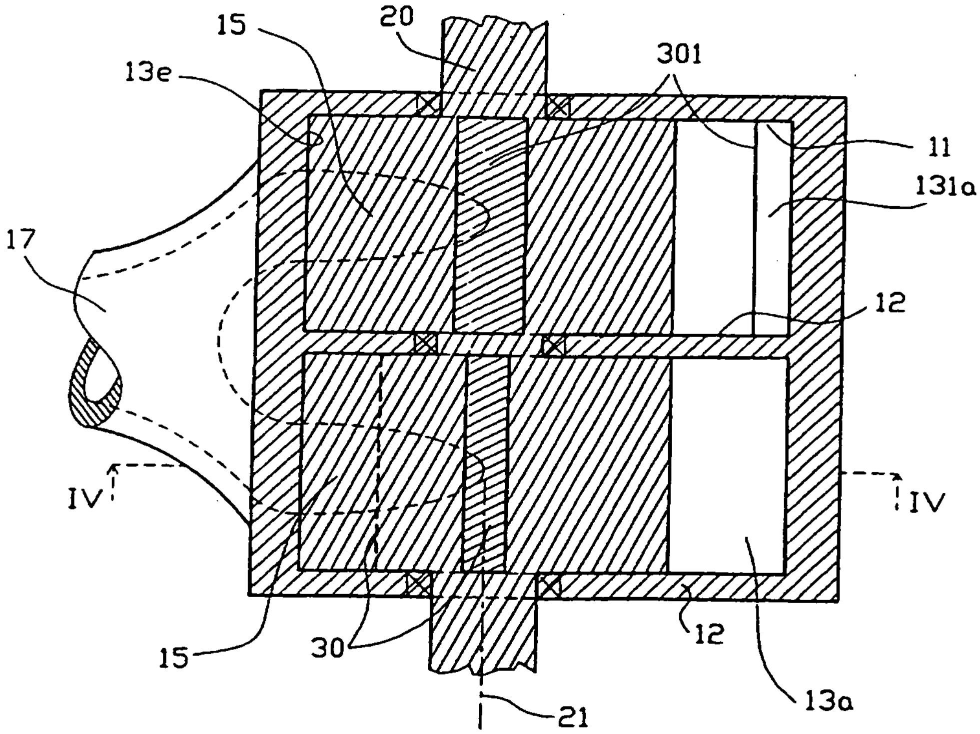

Figure 5

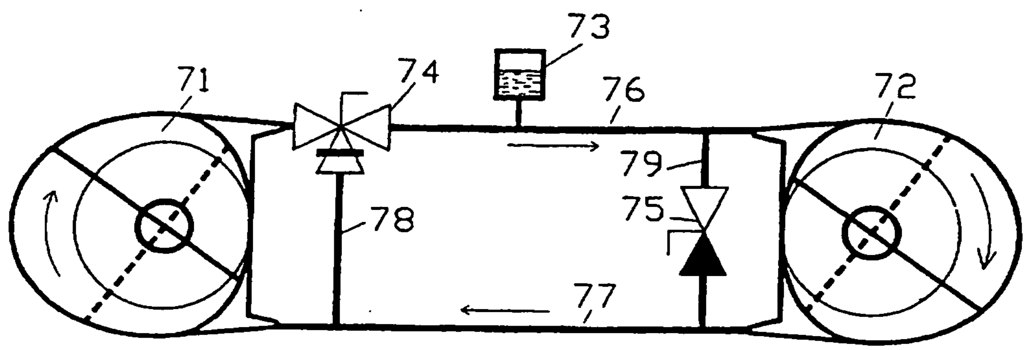

Figure 6

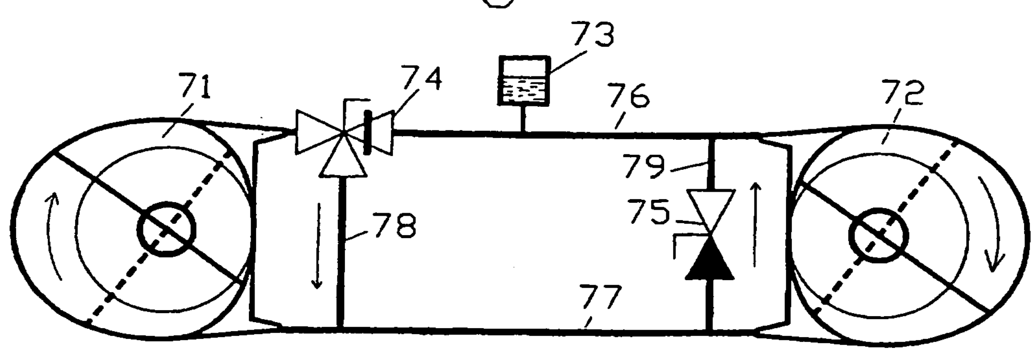

Figure 7

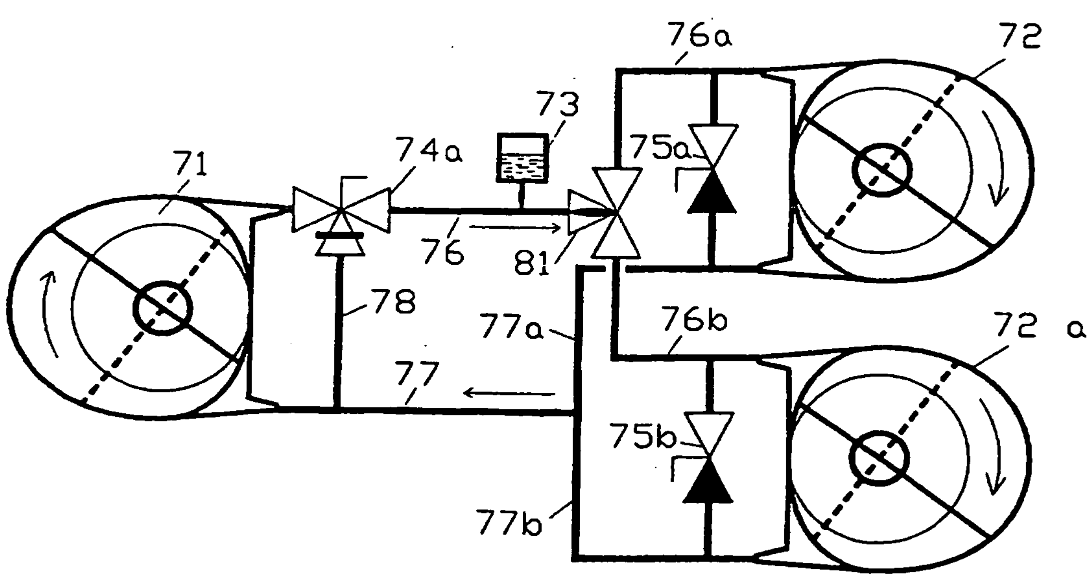

Figure 8

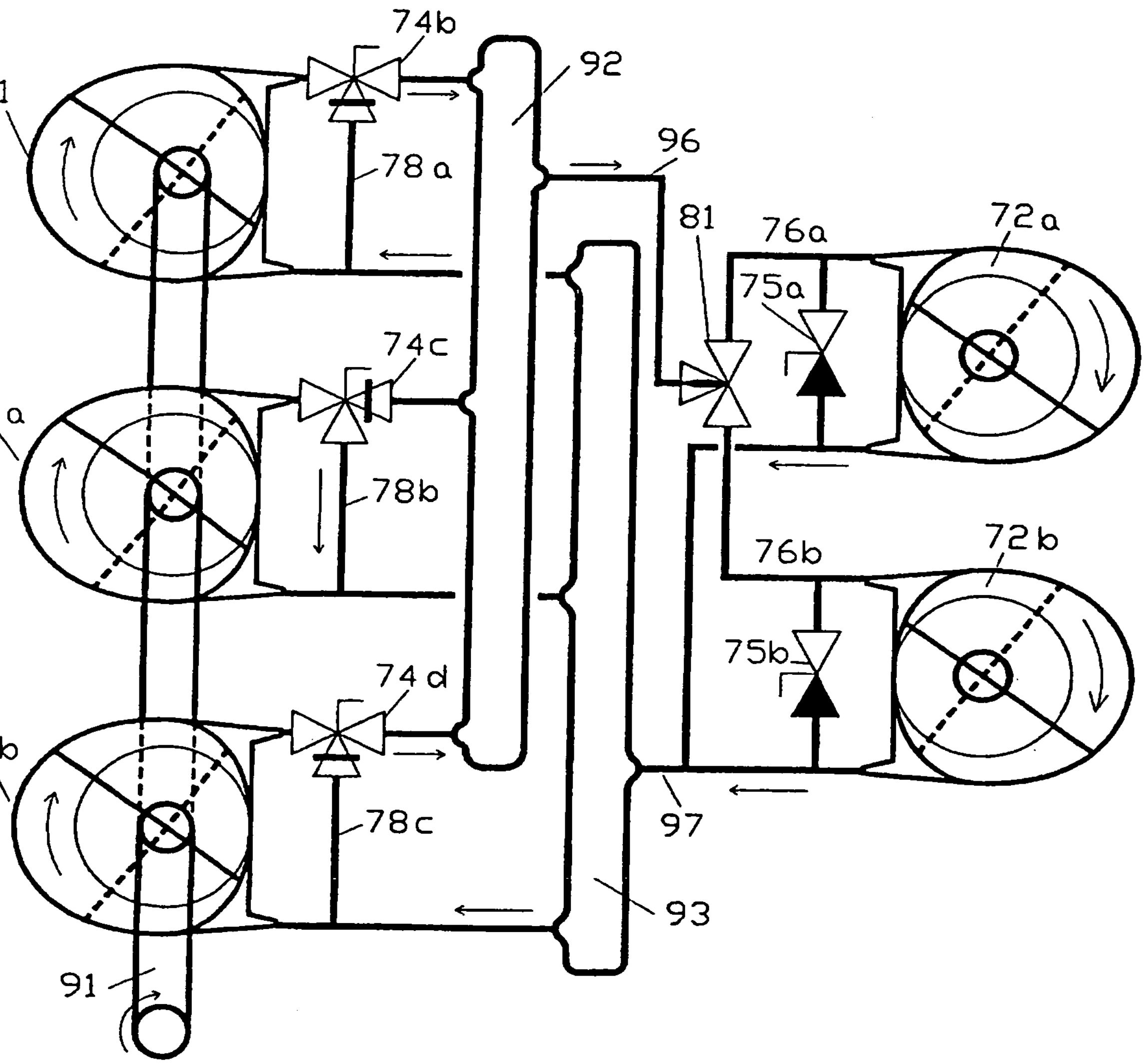
74b
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Figure 9

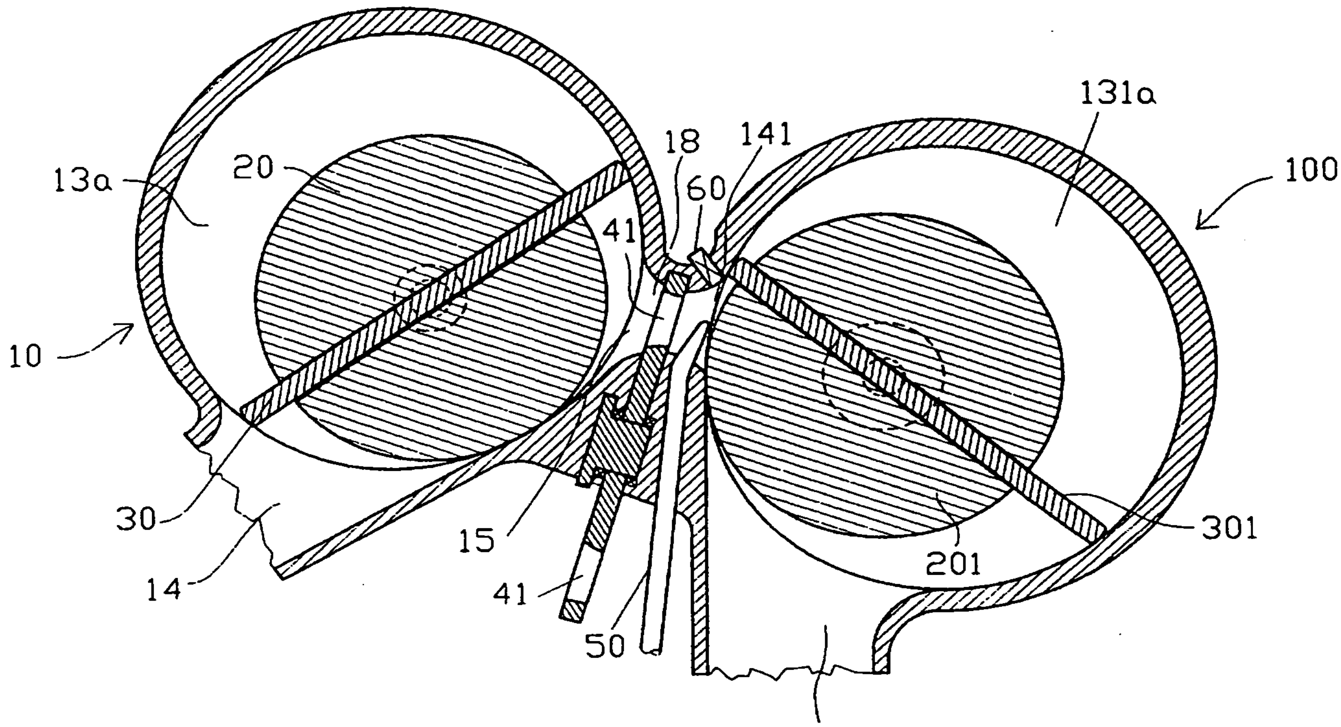

Figure 10

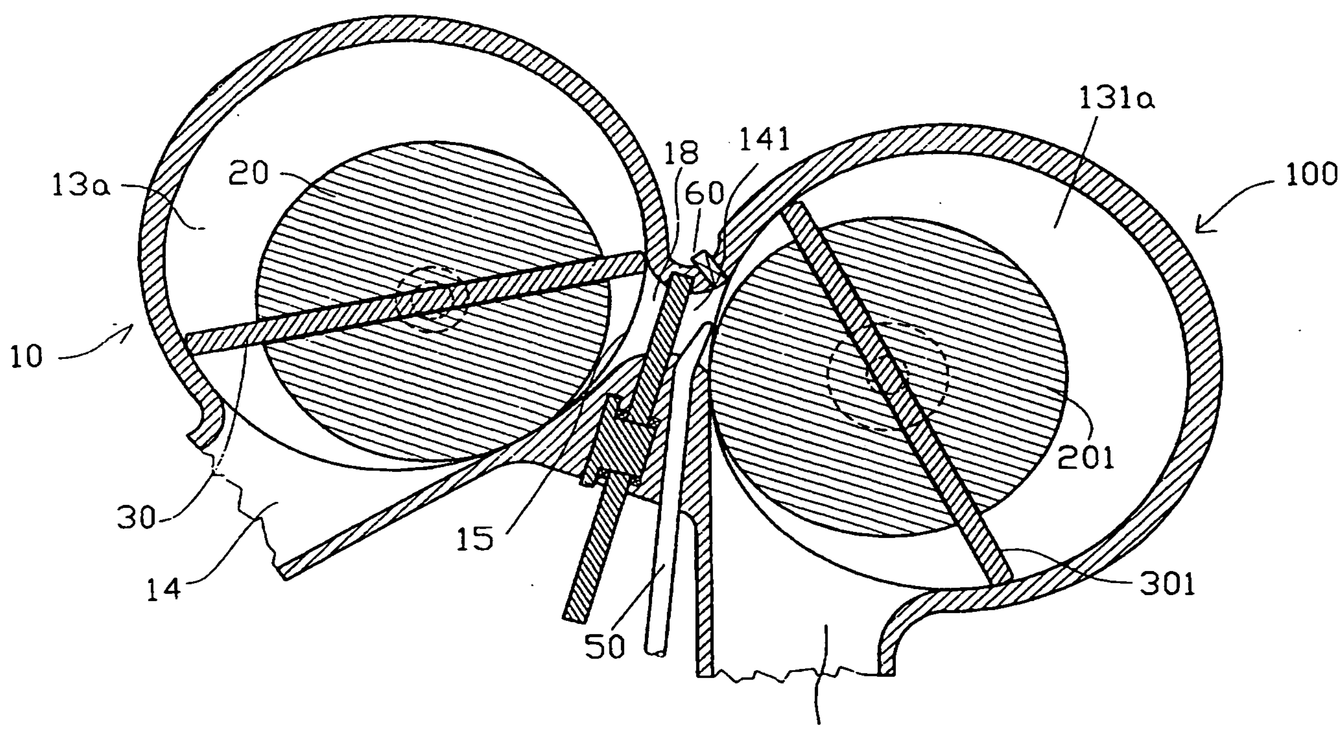

Figure 11

INTERNATIONAL SEARCH REPORT

International Application No
PCT/HU 97/00039

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 F04C2/344

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F04C F01C

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	DE 40 31 468 A (BARMAG) 18 April 1991 see column 2, line 44 - column 3, line 31; figures 1,4	1,3 2,4-7
Y	US 5 006 053 A (SENO) 9 April 1991 cited in the application see column 2, line 31 - column 5, line 18; figures 1-3,5,6	2
X	FR 552 920 A (PAROUFFE) 9 May 1923 see page 1, left-hand column; figures 1,2	1
Y	FR 2 353 729 A (BEPEX CORP.) 30 December 1977 see page 3, line 10 - page 5, line 26; figures	4-6

-/--

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

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"&" document member of the same patent family

1

Date of the actual completion of the international search	Date of mailing of the international search report
24 October 1997	17. 11. 97

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Kapoulas, T

INTERNATIONAL SEARCH REPORT

International Application No
PCT/HU 97/00039

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	DE 42 29 999 A (BRUNS) 10 March 1994 see the whole document ---	7
Y	DE 36 10 703 A (KLAUSNITZER) 21 August 1986 see page 7, line 30 - page 13, line 27; figures ---	7
A	GB 1 139 438 A (GENERAL ELECTRIC) 8 January 1969 see page 2, line 45 - page 4, line 108; figures -----	1,7

1

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INTERNATIONAL SEARCH REPORT

Information on patent family members

Intern. ..al Application No

PCT/HU 97/00039

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
DE 4031468 A	18-04-91	JP 3210087 A	13-09-91
US 5006053 A	09-04-91	NONE	
FR 552920 A	09-05-23	NONE	
FR 2353729 A	30-12-77	AU 2008876 A JP 52076709 A	16-02-78 28-06-77
DE 4229999 A	10-03-94	NONE	
DE 3610703 A	21-08-86	NONE	
GB 1139438 A		NONE	

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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : F04C 2/344

A1

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(71)(72) Applicant and Inventor: ADORJÁN, Ferenc [HU/HU]; Csipke u. 1/b, H-1125 Budapest (HU).

(74) Agent: FABER, Miklós; Advopatent, Fő u. 19, H-1011 Budapest (HU).

(81) Designated States: AU, CA, CH, CZ, IL, JP, KR, NO, PL, RO, SI, SK, US, European patent (AT, BE, CH, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE).

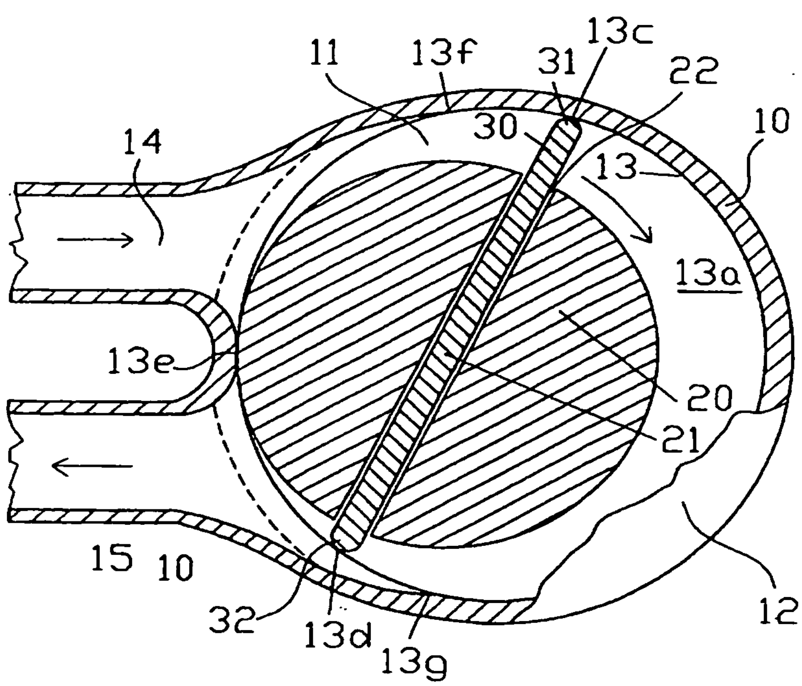
Published
With international search report.
With amended claims and statement.

Date of publication of the amended claims and statment: 5 March 1998 (05.03.98)

(54) Title: ROTARY ASSEMBLY

(57) Abstract

A rotary assembly consisting of a housing (10) with a cavity (13a) that is bounded by the first and the second planar walls (11, 12) as well as by a curved, closed surface (13) in-between, with inlet (14) and outlet (15) openings connecting to the cavity, the rotatably mounted shaft (20) with an axis perpendicular to the planes of planar walls is furnished by a slot (22), parallel and symmetrical to the axis of the shaft and limited within the section of the shaft inside the parallel walls of the cavity and with one or more vanes (30) fitted into the slot which divide the cavity into two or more sub-sections.



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AMENDED CLAIMS

[received by the International Bureau on 19 January 1998 (19.01.98); original claims 1-7 replaced by new claims 1-12 (5 pages)]

1. A rotary assembly, for pumps, compressors, hydraulic or pneumatic motors, and internal combustion engines, comprising of a housing with a cavity, a rotor mounted rotatably to the housing with a fixed axis, the cavity having at least one inlet opening and one outlet opening, said cavity is bounded by a first planar wall and a second planar wall perpendicularly to the axis of the rotor and by at least one cylindrical surface with generatrices parallel to said axis, the circularly cylindrical outer surface of the rotor being in permanent contact with the cylindrical surface of the cavity along at least one common generatrix, the section of the rotor within the parallel planar walls of the cavity supplied by at least one through slot, parallel and symmetrical to the axis of the rotor, one fixed length, rigid vane fitted slidably into each slot, the planar surfaces of the vane perpendicular to the rotor axis being in contact with the planar surfaces of the cavity while the end surfaces of the vane, parallel to the axis of the rotor being cylindrically rounded and being in contact with the cylindrical surface of the cavity, which is characterised by that said cavity being divided at least to two separate chambers (13a,131a), the cylindrical surfaces (13b,131b) of the chambers (13a, 131a) being in contact with the cylindrical surface of the rotor (20) along at least one fixed generatrix (13e,131e) of the chambers (13a, 131a), the chambers (13a, 131a) being separated from each other by an internal planar wall (19) perpendicular to the axis (21) of the rotor (20), each of the chambers (13a, 131a) having one inlet (14,141) and an outlet (15,151) opening, which are on the two opposite sides of the common generatrix (13e,131e), said rotor (20) penetrating the internal planar wall (19) in fluid tight way and being supplied within each of the chambers (13a, 131a) by one through slot (22,221), one sliding vane (30,301) being fitted into each of the slots (22,221), the planar surfaces of the sliding vanes (30,301) towards the internal wall (19) being in contact with on the surfaces of the wall, the sliding vanes (30,301) set to a definite angle relative to each other, in a normal cross section the cylindrical surfaces (13b,131b) of the chambers (13a,131a) being the outer envelope (133a) of a series of the rounding curves (31a,32a) of the end surfaces (31,32) of the vane (30), being fitted to a convex, differentiable equichordal curve (133), the fixed length chords of the equichordal curve (133) passing

through the axis (21) of the rotor, consequently both rounded end surfaces (31,32) of the siding vanes (30,301) being in fluid tight contact with the cylindrical surfaces (13b,131b) of the chambers (13a,131a) along two generatrices (13c,13d) with a constant clearance.

2. A rotary assembly according to claim 1, which is characterised by the rounded end surfaces (31,32) of the sliding vane (30) having generatrices parallel to the axis (21) of the rotor (20), being sections of identical, right, circular cylindrical surfaces. accordingly, in a normal cross section, the envelope (133a) of the equichordal curve (133) of the rounding curves (31a, 32a) being the outer parallel of the equichordal curve (133a) at the distance of the rounding radius (R) of the rounding curves (31a. 32a).

3. A rotary assembly according to claim 1 or 2. which is characterised by the maximum local radius (R_{max}) of the rounding curve (31a, 32a) of the sliding vane (30) with a given width (D) being substantially equal to minimum of the maximal radii (R') determined by the meshpoint of a straight line starting from each point of the equichordal curve (133) so that it is perpendicular to the tangent of the equichordal curve (133) at the given point and the sides parallel to the axis (21) of the sliding vane (30).

4. A rotary assembly according to any of the claims 1 through 3, which is characterised by the inlet opening (14) and the outlet opening (15) of the chamber (13a) extending substantially from the common generatrix (13e) right to the generatrices (13f,13g) determined by the opposite contact generatrices (13f,13g) of the sliding vane (30) at the position perpendicular to the plane determined by the rotational axis (21) and said common generatrix (13e).

5. A rotary assembly according to any of the claims 1. through 4 . which is characterised by the inlet openings (14,141) branching from a common inlet (16) and the outlet openings (15,151) leading into a common outlet (17).

6. A rotary assembly according to any of the claims 1. through 5. which is characterised by the cavity being divided into two chambers (13a,131a), the chambers being equivalent in size and shape and the sliding vanes (30,301) within the two chambers (13a,131a) being perpendicular to each other.

7. A rotary assembly according to claim 6, which is characterised by the cylindrical surfaces (13b,131b) of the chambers (13a,131a) contacting the surface of the rotor (20) along the same common generatrix (13e) and being the chambers symmetrical against the plane (S) determined by the common generatrix (13e) and the rotational axis (21).

8. A rotary assembly according to claim 7, which is characterised by a shape of the cylindrical surfaces (13b,131b) of the chambers ensuring that having a first and a second position of the rotor (20) and the sliding vanes (30,301) with a difference of an acute angle (φ), so that in the first chamber (13a) the contact generatrix (13c) of the first position sliding vane (30) which falling farther away from the common generatrix (13e), as well as the contact generatrix (13c') of the second position of the siding vane (30') which falling also farther away from the common generatrix (13e), being situated on the same side of the symmetry plane (S); the area (A_1) being bounded by the contour line (30a) of the first position sliding vane (30) in the first chamber (13a) starting from said contact generatrix (13c) and continuing on the side facing the second position sliding vane (30'), and by the contour line (30b) on the same side of the second position of the vane (30') also starting from said contact generatrix (13c') and by the cylindrical surface (13b) of the chamber and by the cylindrical surface (20a) of the rotor, and the area (A_2) in the second chamber (131a) being bounded by the contour lines (301a,301b) of the two sliding vane positions (301,301') starting from the contact generatrices (131c,131c'), respectively, being farther away from the common generatrix (13e), determined analogously as in the first chamber (13a), and also by the cylindrical surface (131b) of the chamber and by the surface (20a) of the rotor (20), is proportional to the acute angle (φ) and is independent of the actual first positions of the sliding vanes (30,301).

9. A rotary assembly according to claim 7, which is characterised by a shape of the cylindrical surfaces (13b,131b) of the chambers ensuring, that having a first and a second position of the rotor (20) and the sliding vanes (30,301) with a difference of an acute angle (φ), so that in the first chamber (13a) the contact generatrix (13c) of the first position sliding vane (30) which falling farther away from the common generatrix (13e), as well as the contact generatrix (13c') of the second position of the siding vane (30') which falling also farther away from the common generatrix (13e), being situated on the same side of the symmetry plane (S); the area (A_1) being bounded by the contour line (30a) of the first position sliding vane (30) in the first chamber (13a) starting from said contact generatrix (13c) and continuing on the side facing the second position sliding vane (30'), and by the contour line (30b) on the same side of the second position of the vane (30') also starting from said contact generatrix (13c') and by the cylindrical surface (13b) of the chamber and by the cylindrical surface (20a) of the rotor, and the area (A_2) in the second chamber (131a) being bounded by the contour lines (301a,301b) of the two sliding vane positions (301,301') starting from the contact generatrices (131c,131c'), respectively, being farther away from the common generatrix (13e), determined analogously as in the first chamber (13a), and also by the cylindrical surface (131b) of the chamber and by the surface (20a) of the rotor (20), and the negative difference (A_4-A_3) of the area (A_4) in the same planar cross section of the other end of the first position sliding vane (30) in the first chamber (13a) extending over the cylindrical surface (20a) of the rotor (20), and the area (A_4) determined in the same way for the second position of the sliding vane (30'), is proportional to the acute angle (φ) and is independent of the actual first positions of the sliding vanes (30,301).

10. A rotary assembly according to any of the claims 1 through 4, that is characterised by the outlet (15) of the first chamber (13a) leading to the inlet (141) of the second chamber (131a), the inlet (141) being closable by a movable valve body (40) with an opening (41), and near the inlet (141) there is a fuel injection nozzle (50), and - in given case - an igniting equipment (60).

11. A rotary assembly according to claim 10, that is characterised by having the valve body (40) baing mounted directly on the rotor (20), said valve body (40) being disc shaped and having two symmetrical openings (41).

12. A rotary assembly according to claim 10, that is characterised by the valve body (40) connected to the rotor (20) by a synchronising equipment.

STATEMENT UNDER ARTICLE 19

Referring to the International Search Report dated November 17, 1997, we state, that the characteristic features of our Claims can be found together in none of the following citations:

1	DE 4031 468
2	US 5006 053
3	FR 552 920
4	FR 2353 729
5	DE 4229 999
6	DE 3610 703
7	UK 1139 438

As for the subject matter of our invention:

citations Nos 5 and 7	have no rotor with fixed axis,
citations Nos 5 and 6	have no fixed rigid vane in the slot of the rotor,
citation No 7	has no vane at all,
citations Nos 2 to 7	have no vane with a curved end-surface.

So citations Nos 2 to 7 differ from our invention in the subject matter.

Regarding the characterizing clause:

citations Nos 1 and 2	have no two chambers,
citations Nos 6 and 7	have no internal planar wall between the chambers,
citations Nos 3 to 7	have no envelope curve to the equichordal curve, and
citations Nos 4, 5 and 7	comprise not even an equichordal curve.

The existence of an envelope curve to an equichordal curve is of determining importance, for the operation of such assemblies essentially differs from those without such a shape.

In the same way, an assembly having just one chamber, or two chambers with two different axes are not apt to be part of a multipurpose construction, since in this way the features can not be optimalized.

CORRECTED VERSION*

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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : F04C 2/344 | A1

(11) International Publication Number: WO 98/03794

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(21) International Application Number: PCT/HU97/00039

(22) International Filing Date: 17 July 1997 (17.07.97)

(30) Priority Data:
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(71)(72) Applicant and Inventor: ADORJÁN, Ferenc [HU/HU]; Csipke u. 1/b, H-1125 Budapest (HU).

(74) Agent: FABER, Miklós; Advopatent, Fő u. 19, H-1011 Budapest (HU).

(81) Designated States: AU, CA, CH, CZ, IL, JP, KR, NO, PL, RO, SI, SK, US, European patent (AT, BE, CH, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE).

Published
With international search report.
With amended claims and statement.

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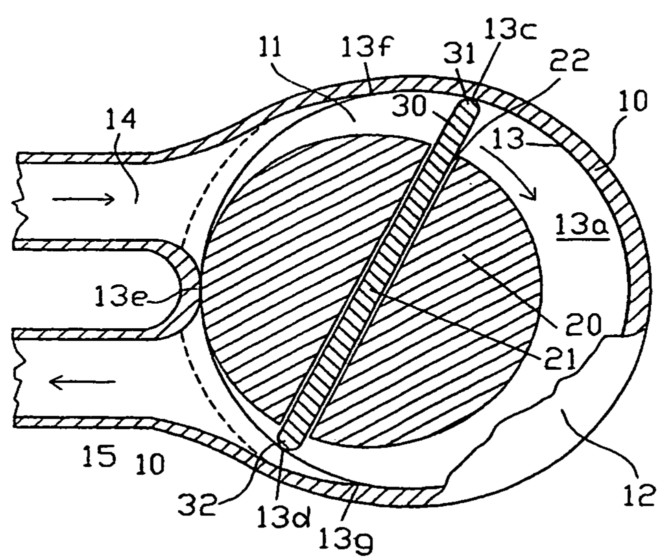
(54) Title: ROTARY ASSEMBLY

(57) Abstract

A rotary assembly consisting of a housing (10) with a cavity (13a) that is bounded by the first and the second planar walls (11, 12) as well as by a curved, closed surface (13) in-between, with inlet (14) and outlet (15) openings connecting to the cavity, the rotatably mounted shaft (20) with an axis perpendicular to the planes of planar walls is furnished by a slot (22), parallel and symmetrical to the axis of the shaft and limited within the section of the shaft inside the parallel walls of the cavity and with one or more vanes (30) fitted into the slot which divide the cavity into two or more sub-sections. The inventive idea will be carried out by means of the following technical measure. The end surfaces (31, 32) of the vane (vanes) toward the curved surface (13) of the cavity are defined by rounded curves in cross section parallel to the planar walls of the cavity, while in any such cross section the curved surface (13) of the cavity follows a non-circular curve, having a primary internal fitting point which lies on the rotational axis (21) of the shaft (20) and this non-circular curve is determined such a way that the rounded end surfaces (31, 32) of the vane (30) are in fluid-tight contact along two opposite generatrices (13c, 13d) with the curved surface of the cavity regardless the angular position of the shaft, the shaft and the internal curved surface of the cavity are also in a permanent fluid-tight contact at least along a single fixed generatrix (13e) or a range of generatrices of the curved surface; while the inlet (14) and the outlet (15) openings are on the two sides of this contact range (13e).



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